Theoretical and Experimental Investigation on the Axial Temperature Mismatch and its Optimization for Coaxial Inertance Pulse Tube Cryocoolers

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ABSTRACT

In a coaxial pulse tube cryocooler, the radial thermal conduction between the pulse tube and the regenerator has a large influence on cryocooler performance. Caused by the temperature mismatch between the pulse tube and regenerator, this phenomenon needs to be carefully considered during the cryocooler design process. In this work, models with and without radial thermal conduction between the regenerator and its coaxial pulse tube have been constructed to analyze the mechanism and predict its effect on cryocooler performance. Experiments have been carried out to characterize the coaxial pulse tube axial temperature distribution under a variety of working conditions. A linear variable differential transformer (LVDT) has been used to analyze the pressure-flow phase angle in the test system. The results were then compared with the simulated results and were used to provide direction for further optimization.

The simulation results show that a steady radial thermal conduction exists between the pulse tube and the regenerator, and this heat transfer can affect the fluid dynamics and thermodynamics in the system. Experiments show that the shape of the wall temperature distribution curve is a useful indicator of the cooler performance and the appropriateness of the arrangement of the pulse tube and the regenerator for achieving optimal working conditions. The simulation results are in good agreement with the experimental data, which verifies the numerical simulation modeling.

To validate the theoretical and experimental studies, the configuration of a previous 2W at 60K experimental prototype PT was redesigned and optimized to get an optimal axial temperature match between the regenerator and pulse tube. The new experimental prototype achieved a COP of 2% at 60 K and 4.3% at 80 K with around 100 W of input electric power.

INTRODUCTION

A pulse tube cryocooler, which has no moving mechanical parts and no displacer seals in the cold head, has the potential to achieve long life, high reliability, less vibration, and lower cost. In recently years, many new theories, such as the enthalpy flow theory and thermoacoustics, have been incorporated to achieve refined thermo-mechanical pulse tube designs such as the double-inlet and inertance tube configurations. These designs have led to cooling capacities and thermal efficiencies comparable with Stirling coolers.[1] These advantages promote the widespread use of the pulse tube cryocooler in many fields, especially in space-borne applications that require a long stable

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lower temperature environment. The coaxial pulse tube configuration, because it has the appearance and utility similar to a Stirling coldfinger, has attracted considerable attention by the cooler development community. However, a drawback of the pulse tube and regenerator being arranged concentrically, is that the different temperature distributions of their inner gases causes radial thermal conduction between the pulse tube and the regenerator. This inherent heat transfer phenomenon has considerable effect on the cryocooler performance.

Radebaugh et al.[2] discussed a simple analytical model and a rigorous numerical model to analyze the regenerator behavior with heat input and removal at intermediate temperature. The simulation and experiments show that the system performance can be improved when heat is removed out of the regenerator. Johnson and Ross[3] built up a heat interceptor between the first and the second regenerator, and Koetttig et al.[4] experimentally studied the immediate thermal heat exchange between the pulse tube and the regenerator on a four-valve U-type pulse tube cryocooler. They also found the same result experimentally. Wang[5] also used a numerical simulation method to analyze the effect of radial conduction on cryocooler performance, and his experiments showed that a pulse tube wall with low conductivity can make the cold end temperature 10K lower.

In this work, numerical simulation models with or without radial conduction between the regenerator and the pulse tube have been built to analyze the heat transfer effect on cryocooler performance. An experimental system has been set up to measure the wall temperature distribution and the pressure oscillation of the working gas in a high frequency coaxial pulse tube cryocooler. The effect of frequency, hot end reject temperature, cooling capacity, and different phase shift mechanisms on the cooler performance and the temperature distribution has been obtained from the experiments. Based on this theoretical and experimental work, a modified pulse tube cryocooler has been designed and fabricated, and it achieved improved cooler performance.

EXPERIMENTAL SETUP

Since the thickness of the wall of the regenerator is very thin (0.25mm) and kept in a vacuum environment, the wall temperature can be approximately considered to be equivalent to the real gas temperature of the working gas. A schematic view of the test apparatus is shown in Figure 1 and a photo of the experimental set up is shown in Figure 2. The test system consists of a coaxial pulse

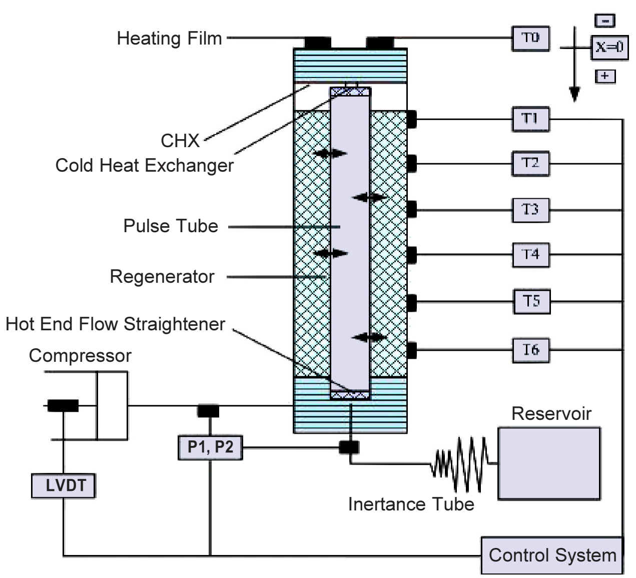

Figure 1. Schematic diagram of the test apparatus with the location of the temperature sensors(T0…T6), pressure sensor and displacement transducer.

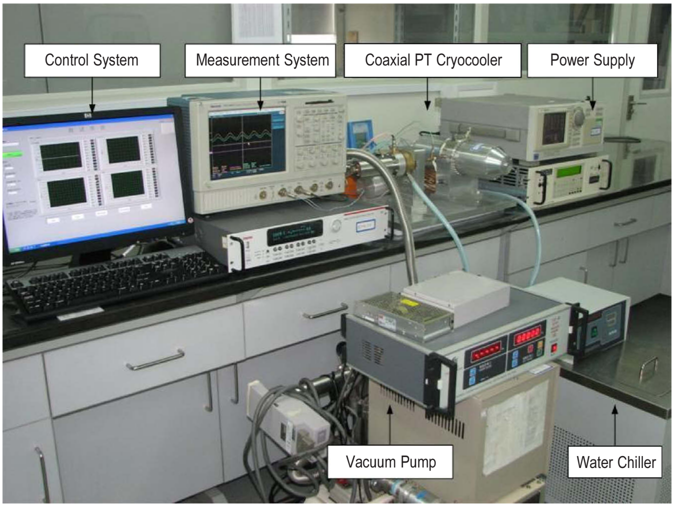

Figure 2. Experimental test set up for axial temperature measurement

tube cryocooler, the measurement system, and the control system. The coaxial pulse tube is one of our first-generation 60K at 2W experimental prototypes.[6] The aftercooler of the pulse tube cryocooler is kept at a certain temperature with circulating cooling water. A well designed split Oxford-type linear compressor with maximum 8.2cc swept volume is used to drive this cold finger. Two inertance tubes with different inner diameter and length together with the reservoir are used as the phase shift mechanism. A resistive heating load was used to supply heat to the cold head. A calibrated LakeShore Cernox Resistance Sensor (Lakeshore CX-63922) was used to measure the cold tip temperature, and another six PT100 thermometers were uniformly arranged along the wall of the regenerator. Two pressure probes were mounted at the entrance of the regenerator and at the hot end of the pulse tube. An LVDT sensor was used to monitor the compressor piston movement, which can help to estimate the PV power transferred from the compressor piston face to the cold finger. Labview software on a personal computer was used to control the measured apparatus and to acquire, display, and process the test data.

NUMERICAL SIMULATION ANALYSIS

We have conducted numerical simulation modeling of the pulse tube cryocooler based on a CFD model[6] developed in our laboratory. The dimensions and phase shift mechanisms were also simulated and optimized by the model. In order to investigate the effect of heat transfer on the cryocooler performance, we constructed two models: the first was of an in-line configuration of an annular regenerator with no thermal link to the pulse tube, whereas the second was for a concentric arrangement of the regenerator and pulse tube that considered radial conduction. Both were used to analyze the effect of radial thermal conduction on the cooler performance.

In the models, the temperature of the hot end was held at 300K and that of the cold head was 60 K. When the piston amplitude in the simulation models is 3.5 mm, the corresponding PV power in the compression space is about 75W. The simulated cryocooler performance is shown in Table 1.

Table 1. Comparison of the numerical simulation results of the in-line type and coaxial type pulse tube cyocooler

Model type	PV power(W)	Pr1	Pr2	$\dot{m}_h$ (g/s)	$\dot{m}_c$(g/s)	Phase1(deg)	Qc@60K (W)	COP %
Linear type	75.300	1.196	1.132	2.650	2.910	27.000	2.509	3.330
Co-axial type	74.430	1.174	1.131	3.230	2.920	24.000	2.309	3.100

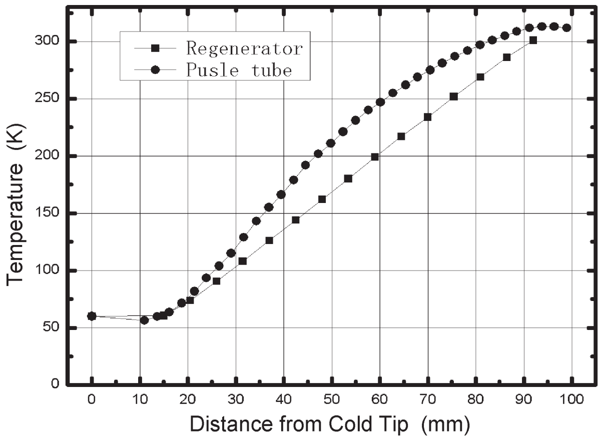

Figure 3. Numerical simulation of the axial gas temperature distribution of the pulse tube and the regenerator without thermal conduction between the regenerator and the pulse tube

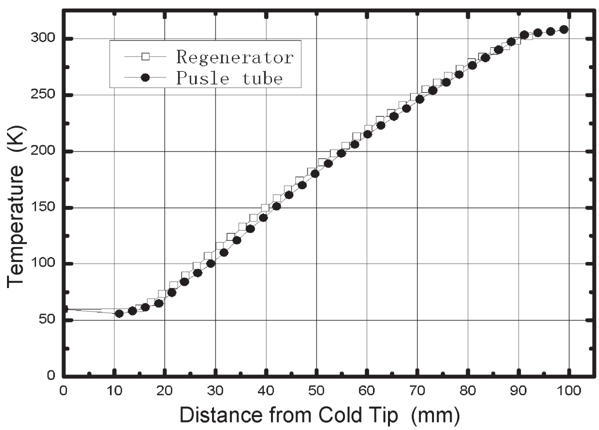

Figure 4. Numerical simulation of the axial gas temperature distribution of the pulse tube and the regenerator with thermal conduction between the regenerator and the pulse tube

As shown in Table 1, the linear coldhead achieved better performance than the coaxial configuration. The increase of the pressure ratio (Pr1, Pr2) at the compression space and the cold end of the regenerator, the better phase shifts (Phase1) between the mass flow and the pressure wave at the cold end of the regenerator, and the decreased mass flow ($\dot{m}_h$, $\dot{m}_c$) all result in lower irreversible entropy-generation, which can account for the decline of the performance.

Figure 3 shows the numerical simulation results for the gas temperature distribution curve of the linear arrangement. As shown, the cold end temperature of the gas in the pulse tube is lower than the cold end gas temperature in the regenerator, while higher at the hot end. The axial gas temperature distribution of the regenerator approximates the ideal linear distribution, while the pulse tube temperature distribution curve is convex in the middle part. This temperature difference, if the two were conductively coupled along there lengths, would cause thermal transfer between the gas in the pulse tube and that in the regenerator. This would result in an irreversible heat transfer loss and deteriorate the cryocooler performance.

Figure 4 shows the axial gas temperature distribution in the coaxial configuration. As shown, the gas temperature of the pulse tube and the regenerator are almost the same. A steady radial thermal conduction exists between the pulse tube and the regenerator. The shape of the temperature

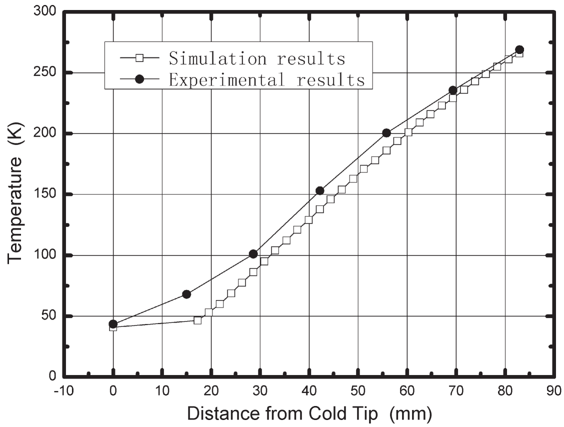

Figure 5. The numerical simulation and the experimental results of the axial temperature distribution

curve is approximately linear with just a little convex bend. The altered shape is similar to the pulse tube temperature distribution curve simulated in the linear PT model. This again demonstrates that the regenerator gas is being heated by the working gas in the pulse tube through the thin wall of the pulse tube.

EXPERIMENTAL RESULTS AND DISCUSSION

In the experiments, wall temperature distribution curves and phase shift changes were obtained for a coaxial IPTC for a variety of operating frequencies, hot end rejected temperatures, phase shifting mechanisms, and cooling capacities at the cold end. All the experiments were performed with a 3.2 MPa charging pressure and an input electric power of 120 W.

Comparison between the Numerical Simulation and the Experimental Results

In order to validate the correctness of the numerical simulation model with respect to the radial thermal conduction, corresponding results have been selected from the simulation and experiments, and a comparison has been made as shown in Finger 5. The experimental results were obtained with an input electric power of 120W and a hot end reject temperature of 10°C. The corresponding PV power in the compression space was 63.9 W calculated from the measurement apparatus. The simulated results from the model were obtained using a 3.2 mm amplitude for the piston and a corresponding PV power of 62.36 W. As Figure 5 shows, the performance and the temperature distribution curves are approximately consistent with each other. That the experimental results are higher than the simulated ones may be because of the heat transfer temperature difference through the wall of the regenerator and the thermal contact resistance in the measuring process.

Effect of Operating Frequency on the Axial Temperature Distribution

Figure 6 shows the wall temperature distribution curves for steady conditions with various operating frequencies. The shape of the curves is convex, which is consistent with the cold finger temperature distribution curve simulated above. The dark shaded symbols indicate the optimal frequency of 55 Hz, which will be set as the reference line for analysis.

Figure 7 shows the phase shifts between the volume flow and the pressure wave at the piston face. The phase shifts are seen to increase as the frequency increases. As Figures 6 and 7 show, the axial temperature distribution deviates from the optimized frequency curve, and the phase shifts become worse as the frequency departs from the optimal frequency. The average phase shifts between the mass flow and the pressure wave become higher in the regenerator, which can increase the flow resistance loss and make the performance worse.

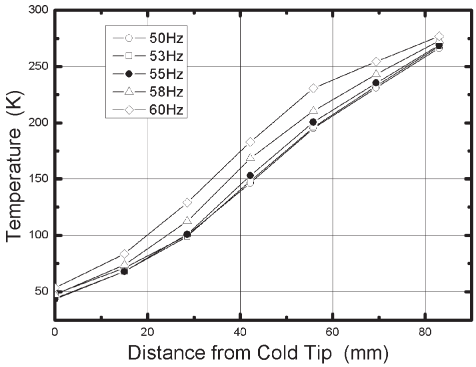

Figure 6. Wall temperature distributions curves versus operating frequency

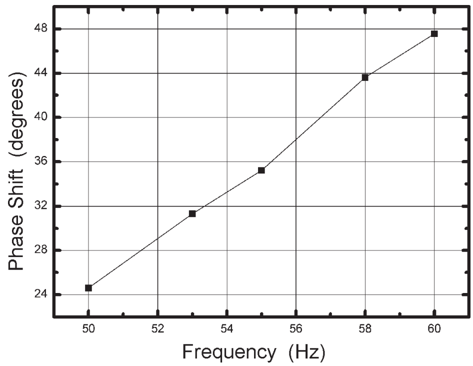

Figure 7. Phase shifts between volume flow and pressure wave versus operating frequency

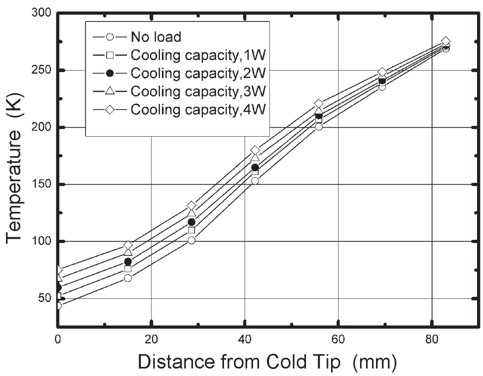

Figure 8. Axial temperature distribution versus cooling capacity

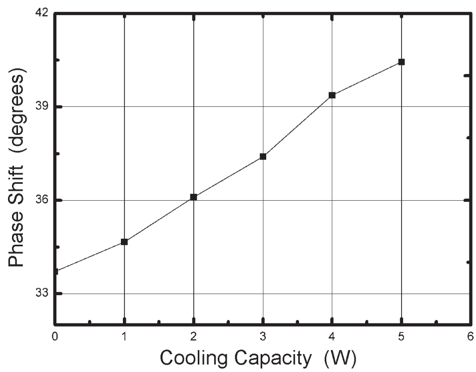

Figure 9. Phase shifts between volume flow and pressure wave versus cooling capacity

Effect of Cooling Capacity on the Axial Temperature Distribution

Figure 8 shows the experimental results of the dependence of the wall temperature profiles of the cold finger on the cooling capacity, and Figure 9 shows the corresponding phase shifts between the volume flow and pressure at the compression space. As they show, the cooling capacity only affects the slope coefficient of the axial temperature distribution, and the phase shifts increase as the cooling capacity increases.

Effect of Reject Temperature on the Axial Temperature Distribution

Figure 10 shows the effect of the reject temperature on the axial temperature distribution. As it shows, the reject temperature can change the slope of the distribution of the wall temperature. The higher the reject temperature, the more steep the temperature curve is , which can increase the axial conduction loss through the cold finger. Figure 11 shows the phase shifts between the volume flow and pressure at the compression space. The phase shift doesn't vary significantly as the hot end reject temperature changes from 10°C to 25°C.

Effect of Phase Shifting Mechanisms on the Axial Temperature Distribution

The simulation results show that the optimal operating frequency of the cold finger can be easily shifted, while getting the same cold finger efficiency, by only making changes to the inertance tubes dimensions. This provides a very convenient means to alter the operating frequency of the cold finger to match the optimal frequency of the driving compressor. In this paper, three groups of inertance tubes making the cold finger work at frequency of 50Hz, 55Hz, 59Hz have been

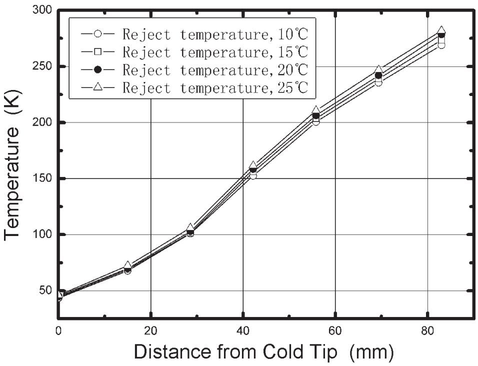

Figure 10. Axial temperature distribution versus hot end reject temperature

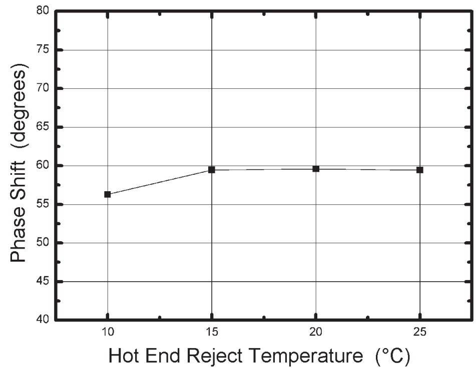

Figure 11. Phase shifts between volume flow and pressure wave versus hot end reject temperature

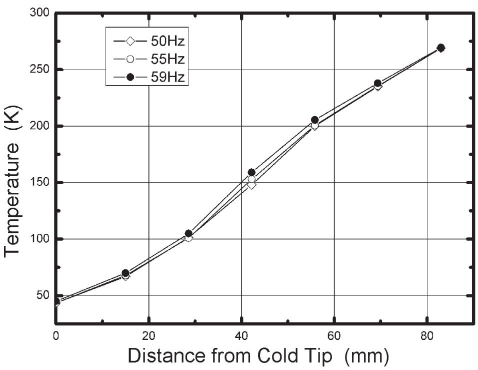

Figure 12. Axial temperature distribution curves versus different phase shift mechanisms

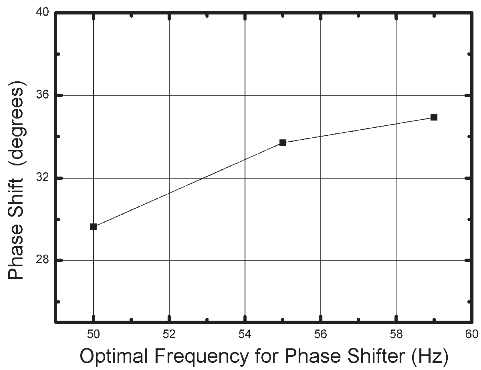

Figure 13. Phase shifts between volume flow and pressure wave versus different phase shift mechanisms

designed and fabricated. Figures 12 and 13 show the cold finger axial temperature distribution curves and the phase shifts with different phase shift mechanisms. The three groups had almost the same performance and the same axial temperature distribution. The optimal phase shift increases as the optimal frequency increases. The unchanged compressor efficiency may account for the steady cryocooler performance.

PERFORMANCE INVESTIGATION OF A NEW COAXIAL PULSE TUBE COOLER

Based on the above theoretical and experimental work, a new pulse tube cooler was designed with the pulse tube and the regenerator configurations rearranged to create an optimal axial temperature match. The phase shift mechanisms were also redesigned to realize an optimal phase shift.

Figures 14 and 15 show a representative cool down curve and cooling capacities achieved for different input power levels. The cooler can obtain a cold end temperature below 60 K in about 8 minutes, and a lowest temperature of 42 K can be reached in less than 18 minutes. A cooling capacity of 2 W at 60 K and 4.47 W at 80 K can be obtained with an input electric power of 104 W at 300 K hot end reject temperature.

SUMMARY AND DISCUSSION

The Numerical simulation model predicts that the axial temperature mismatch has a considerable effect on the cooler performance. The wall temperature distribution curve can be regarded as an experimental judgement standard for whether the operating parameters and phase shift mecha-

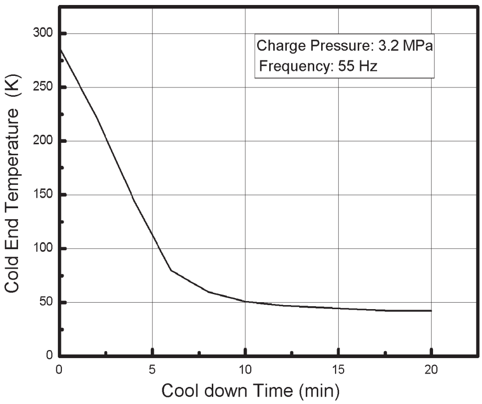

Figure 14. Cool down curve of the newly designed coaxial pulse tube cryocooler.

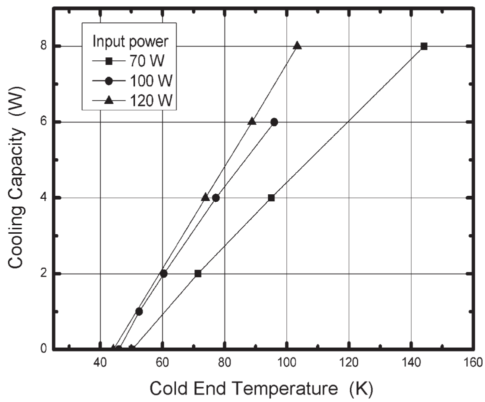

Figure 15. Cooling capacity versus cold end temperature.

nism are appropriate. This is also verified by experimental data. The theoretical prediction and experimental measurements are in good agreement with each other. A newly designed version considering the axial temperature match between the pulse tube and the regenerator has achieved 2W at 60 K and 4.47 W at 80 K with an input electric power of 104W, with the hot end reject temperature of 300 K. The following activities are being considered to obtain further improvements:

1. More rigorous simulation of the actual heat transfer between the regenerator wall and the inner gas and matrix will be done. This is motivated because the thickness of the wall and the heat exchanger temperature of the cold and hot ends can affect the temperature distribution, and thus make the temperature depart from the real gas temperature.
2. More theoretical and experimental activities will be done on pulse tube cryocoolers with different dimensions and different phase shifting mechanisms.

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